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PERFORMANCE ESTIMATION OF HERMETIC RECIPROCATING COMPRESSOR WITH COMPUTER MODEL

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ABSTRACT

This paper describes a whole simulation model with personal computer for estimating the performance of hermetic reciprocating compressors used in refrigerators. The computer package includes six control volumes and three major calculating modules, which contain simulations of valves dynamics, compressor thermodynamics and mechanisms. From the frictional loads and indicated work that were derived from the thermodynamics and mechanism simulation procedure respectively, can evaluate the motor efficiency, volumetric efficiency, adiabatic compression efficiency and mechanical efficiency, and then the energy efficiency ratio (E.E.R.) of the reciprocating compressor can also be obtained.

In this study, the valve dynamic, which uses with a nonlinear mode to deal with the motion of the reed valve, thermal and mechanism analysis are incorporated to simulate the compressor performance. The E.E.R. of the compressors shows well agreement between simulation predictions and experimental data in two different valve thicknesses and the feasibility of this computer model also has been verified.

1. INTRODUCTION

The hermetic reciprocating compressors are used as cooling facilities, refrigeration and air conditioners in the mechanical, material or chemical industries widely. Many papers have presented the overall performance of reciprocating compressor. Zhou and Hamilton (1986) simulate multi-cylinders reciprocating compressor, they set three control volumes, suction chamber, discharge chamber and cylinders, and assume the refrigerant inside each control volume are uniform. Their derived equation of state is gotten by using compressible coefficient to modify ideal gas. Morel and Keriber (1988) develop a comprehensive reciprocating compressor simulation, which includes efficiency analysis, piston/cylinder fraction, blow-by, valve flow and dynamics. Cavallini *et al.* (1998) presents an unsteady-state analysis of compression cycle, which allows a direct computation of the heat, work and mass flow rates. Huang and Chen (2001) introduce the reciprocating compressor's volumetric and adiabatic efficiency with assuming six control volumes in a compressor and reed valve motion model as a linear model. Hsiung and Tseng (2002) simulate reciprocating compressor performance by integration of thermal and mechanism analysis.

This study combines the valve dynamic, thermal and mechanism analysis to simulate the reciprocating compressor performance with calculating volumetric efficiency, adiabatic compression efficiency, mechanical efficiency, motor efficiency and E.E.R.. The E.E.R. of simulation results is in good agreement with experimental data in two cases of different valve thickness. And the feasibility of this developed simulation program has been verified.

2. STRUCTURE OF SIMULATION PROGRAM

The passage of the refrigerant flowing through a reciprocating compressor, from the suction pipe to discharge pipe, comprises many components such as suction port, suction plenum, suction muffler, suction valve, cylinder, discharge valve, discharge muffler and discharge port etc. (see Figure 1). In this study, six control volumes were assumed in a compressor (see Figure 2) and the properties of the refrigerant in the same control volume were assumed as uniform. A real gas simulation of HFC-134a was used for the calculation of the compressor performance.

The structure of the simulation package is presented as showed in Figure 3 and it contains five main modules: parameters input, valve simulation, thermodynamics, dynamics balance and efficiencies output. Every module includes a specific calculating kernel that couldn't work independently.

The valve simulation generates the suction and discharge valve models that thermodynamics uses. Next, the thermodynamic simulation module calculates the mass flow rate, indicated work, gas loads, or other thermodynamic parameters that the kinematics and dynamic force analysis of the compressor body needs. Then, the program evaluates the bearing performance and also calculates the total frictional and lubrication loads on the shaft and moving parts.

After the friction power loss of mechanical analysis is acquired and the input power of motor can be obtained by the experiment data in a dynamometer, the updated parameters are passed back to the slider-crank mechanism analysis to perform the next step kinetic and dynamic forces analysis. The iteration calculating doesn't converge until the percentage error of total power loss of motor is smaller than a specified value.

Finally, the program calculates volumetric efficiency, compression efficiency and mechanical efficiency of the developed compressor, and then the energy efficiency ratio (E.E.R.) can be evaluated.

The following equations show the calculation of the compressor efficiencies and E.E.R. (McQuiston *et al.*, 2000)

$$h_v = 1 + C - C \left(\frac{P_d}{P_s} \right)^{1/n} \quad (1)$$

where h_v is the volumetric efficiency, C is the clearance factor that is the ratio of the clearance volume to the piston displacement, P_d is discharge pressure of the cylinder, P_s is suction pressure of the cylinder, n is the polytropic exponent which is determined experimentally.

$$h_{ad} = \frac{w_{th}}{w_{actual}} \quad (2)$$

$$w_{th} = \frac{n}{n-1} P_s v_s \left[\left(\frac{P_d}{P_s} \right)^{(k-1)/k} - 1 \right] \quad (3)$$

where h_{ad} is the adiabatic compression efficiency, w_{th} is isentropic compression work, k is isentropic exponent, w_{actual} is actual compression work.

$$h_m = \frac{W_{actual}}{W_{actual} + W_{mechanical}} \quad (4)$$

where h_m is the mechanical efficiency, W_{actual} is actual compression power which is defined as the multiplication of the refrigerant mass flow rate and the actual compression work, $W_{mechanical}$ is the mechanical power loss that can be derived from the mechanism simulation analysis.

$$h_{motor} = \frac{W_{actual} + W_{mechanical}}{W_{motor}} \quad (5)$$

where h_{motor} is the motor efficiency, W_{motor} is motor input power that is obtained from the experimental data in a dynamometer.

$$E.E.R. = \frac{Q_c}{W_{motor}} \quad (6)$$

where $E.E.R.$ is energy efficiency ratio, Q_c is capacity of the compressor, W_{motor} is motor input power.

3. THEORETICAL ANALYSIS OF VALVE

The opening conditions of the cylinder valves seriously influence the working efficiency of refrigerant in the cylinder, especially for the volumetric and compression efficiency. In this study, we use nonlinear theory to simulate the motion of the reed valve, because the displacement of reed valve is several times to the thickness of the valve used in our hermetic reciprocating compressor. A linear model can't provide accurate results if the huge amplitude is involved in the actual valve vibration (Sheu and Hu, 2000). Different valve, port and seat geometries, such as thickness, equivalent length etc., affect the effective flow area and the effective force area, which are important parameters for the vibration analysis of the reed valve and the compressor's performance simulation. Ferreira *et al.* (1986, 1988) provided the experimental data of the effective flow and force distributions, also has been used in this study.

The Lagrangian approach is used to derive the governing equation of motion for a cantilever type valve and the assumed-modes method is used to simplify the governing equations of the reed valve vibration. The valve displacement, $u(x,t)$, is assumed as a solution of the boundary-value problem associated with a conservative continuous system. The valve displacement, which is depicted in Figure 4, is expressed as following.

$$u(x,t) = f(x)q(t) \quad (7)$$

where $f(x)$ is valve shape function (or valve admissible function), $q(t)$ is the time-dependent generalized coordinate.

In the case of the suction valve, F , which is generalized force acting on the valve, is expressed as follows (Sheu and Hu, 2000).

$$F = \mathbf{b}f(l_1)(p_s - p)\left\{r_s^2 + [f(l_1)]^2 q^2\right\}/m \quad (8)$$

where \mathbf{b} is the effective force area coefficient which is related with the valve displacement $u(x,t)$ and obtained from experiments (Ferreira *et al.*, 1986, 1988), l_1 is the distance of the applied force related to the coordinate, p_s is pressure in the suction chamber, p is pressure in the cylinder, r_s is radius of the suction orifice, m is equivalent mass of the valve.

The pressure field inside the cylinder can be expressed as

$$\frac{dp}{dt} = \frac{\mathbf{a}A_v k p_s}{V} \left(\frac{p}{p_s}\right)^{\frac{1}{k}} \sqrt{\frac{2kRT_s}{k-1} \left[1 - \left(\frac{p}{p_s}\right)^{\frac{k-1}{k}}\right]} - \frac{k p}{V} \frac{dV}{dt} \quad (9)$$

where the valve effective flow area coefficient \mathbf{a} is determined from experiment and is a function of the displacement of the valve (Ferreira *et al.*, 1986, 1988), A_v is flow area of the suction valve, k is isentropic exponent, p is pressure inside the cylinder, p_s is pressure in the suction chamber, V is instantaneous volume of the cylinder, R is gas constant, T_s is suction temperature.

The flow through the valve port is assumed to be one-dimensional adiabatic flow. The effective flow area is varying by the valve lift, and decided by experiment. By assuming the flow direction is $1 \rightarrow 2$ (e.g. suction chamber to cylinder), the flow through the valve port is isentropic, and ignore potential energy, the mass flow rate may be calculated.

$$\frac{dm}{dt} = \mathbf{a}A\sqrt{2r_1(p_1 - p_2)} \quad (10)$$

where m is the refrigerant mass flow through the valve port, the valve effective flow area coefficient \mathbf{a} is determined from experiment and is a function of the displacement of the valve, r_1 is refrigerant density upstream, p_1 and p_2 are upstream and downstream pressure respectively. The effective flow area is defined as the multiplication of the effective flow area coefficient, \mathbf{a} , and the valve orifice area A .

4. RESULTS AND DISCUSSION

Applying this developed technique, two kinds of compressors, which are classified by two suction valves with thickness 0.15 mm and 0.20 mm, were analyzed and experiments in a calorimeter with ASHRAE test condition in

order to compare the E.E.R. and verify the feasibilities. The calorimeter measuring accuracy is 96% for energy efficiency ratio (E.E.R.).

All the conditions and parameters used in analysis and experiments are identical as Table 1 and Figure 1 shown. The E.E.R results from the software simulated in different valve thickness are 1.3093 and 1.2773, which are close to the experiment results (E.E.R = 1.296 and 1.275) respectively. .

Figure 5 shows the effects of different valve thickness on the valve lifting and the time interval of valve lifting. A thinner valve such as 0.10 mm causes very large valve displacement and much longer vibration transient time, while the thicker valve such as 0.25 mm reduces the displacement of the valve lifting, delays the valve sealing to the seat, and makes more refrigerant flows back through the suction orifice in suction process, meanwhile, the refrigerant mass flow rate reduces such as Figure 6 evaluated

The effects of the valve thickness on the compressor compression efficiency are depicted in Figure 7. Although the thinner valve has good performance in flow dynamics in the cylinder and hence enlarges the effective force area, damage may happen and shorten the valve reliability. The valve of 0.15 mm thickness had broken after we performed the lift testing experiments of the valve. Therefore, the selected valve thickness in this study is 0.20 mm that can compromise the compressor efficiency and working life to optimum

5. CONCLUSIONS

A performance simulation package of reciprocating compressor with window-based, was developed by using Borland Built C++ and built in the object-oriented programming (OOP) concept, which could work with friendly and obviously. Applying this program, the volumetric efficiency, compression efficiency, mechanical efficiency, motor efficiency and the energy efficiency ratio (E.E.R.) of the developed reciprocating compressor, can be evaluated easily.

The computer calculating results show well agreement between simulation predictions and experimental data with various valve thicknesses, meanwhile, and the feasibility of this computer package has been verified. This package has been used for the development of the reciprocating compressor which is applied in household refrigerator system.

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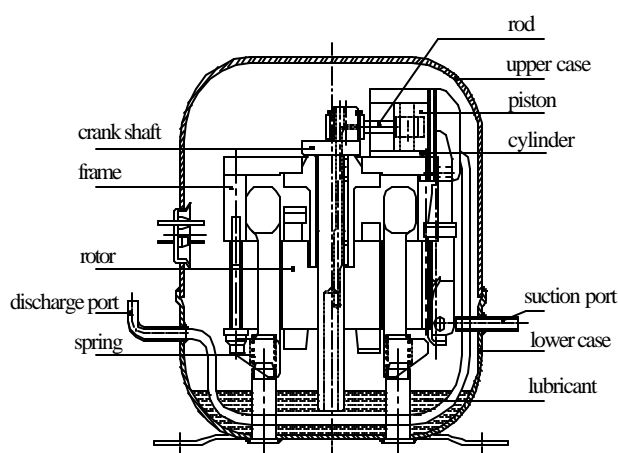
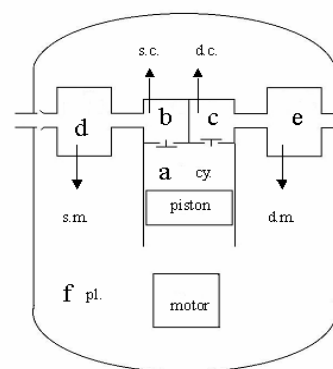


Figure 1: The schematic outline of the developed compressor



a:cylinder; b:suction chamber; c:discharge chamber;
d:suction muffler; e:discharge muffler; f:suction plenum

Figure 2: The divisions of six control volumes used in this study

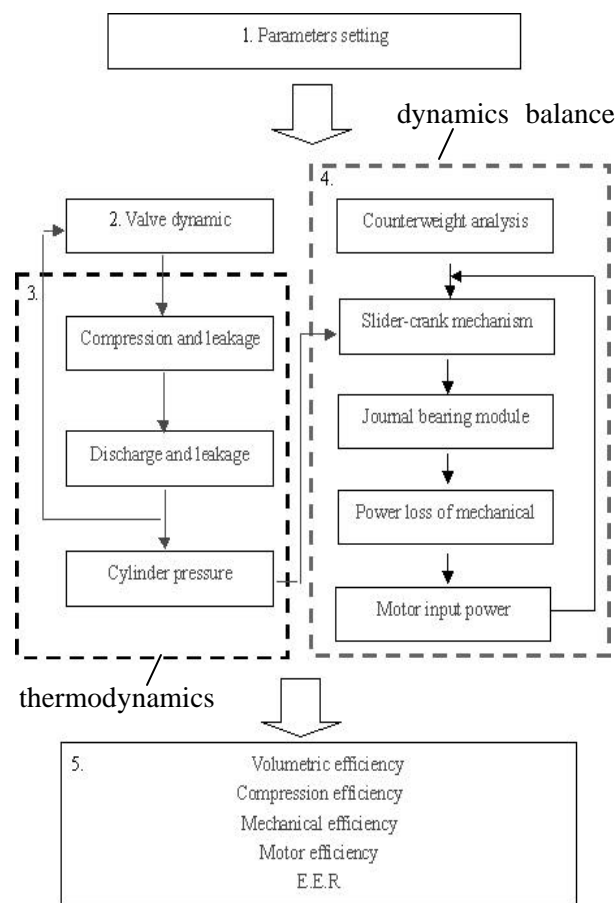


Figure 3: The structure of the simulation package used in this study

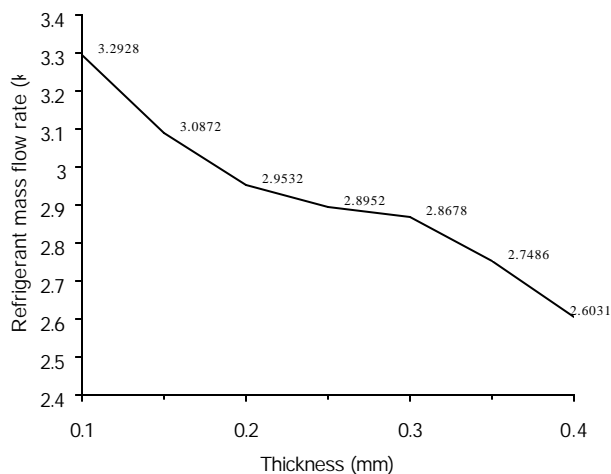


Figure 6: Refrigerant mass flow rate vs. suction valve thickness

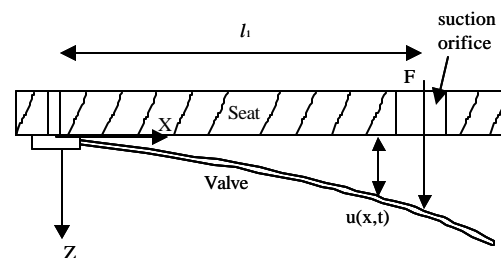


Figure 4: Suction valve displacement, $u(x,t)$, used in this study

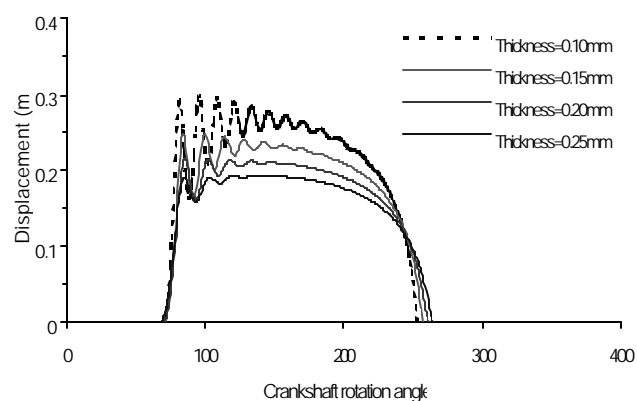


Figure 5: Suction valve displacement vs. thickness

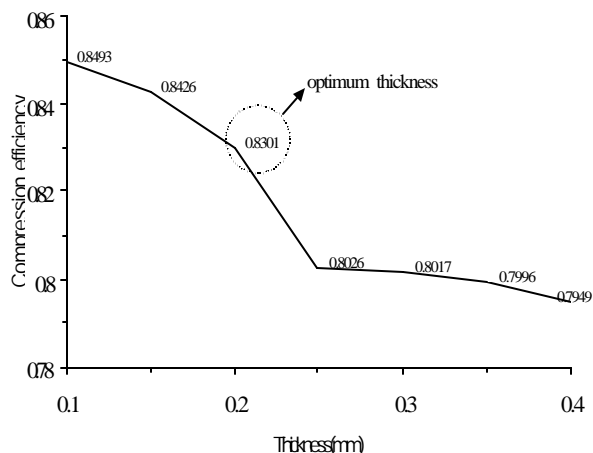


Figure 7: Compressor compression efficiency vs. suction valve thickness

Table 1: Geometric specifications of the reciprocating compressor used in this study

Cylinder diameter	25 mm
Length of rod	38.5 mm
Eccentric radius of rod	7.5 mm
Suction chamber	24.25 cm ³
Discharge chamber	22.42 cm ³
Suction muffler	24.25 cm ³
Discharge muffler	22.42 cm ³
Lift limit of suction valve	2 mm
Lift limit of discharge valve	1 mm
Diameter of suction orifice	4 mm
Diameter of discharge orifice	4 mm
Refrigerant & Oil	HFC-134a; POE VG22

Table 2: Compressor efficiencies & E.E.R. between simulation and experiment results

Suction valve thickness	0.15mm	0.20mm
Motor rotational speed	3600rpm	3600rpm
Volumetric efficiency	76.1%	74.3 %
Compression efficiency	84.26%	83.01 %
Mechanical efficiency	58.97%	58.17 %
Motor efficiency	74.87%	74.66%
E.E.R (simulation results)	1.3093	1.2773
E.E.R (experimental results)	1.296	1.275
Deviations in E.E.R.	1.03%	0.18%